CHAPTER 2

DESIGN CONSIDERATIONS

2-1. General.

a. To fulfill the basic objectives of the prime mission material/personnel (PMMP), the HVAC system must prevent CB contamination and ensure survival and operation of occupants and equipment in a degree consistent with the other elements of the facility as required by the PMMP or specified by facility designers or both. The methodology for integrating the HVAC design in the total design of an effective facilities is covered in TM 5-858-1 in the context of availability, survivability, endurance, performance, technical and cost effectiveness, trade offs and optimization, and functional and interface compatibility. HVAC includes both nonprotective and protective elements, such as hardened structures, reservoirs, tunnels, and penetrations.

b. In particular, the hardened air-entrainment subsystem (AES), which ensures the continuous or periodic transfer of air between the atmosphere and the facility, is covered in TM 5-858-5. The AES design includes ports, ducts and chambers, blast valves, dust removal devices, and booster blowers. The TM 5-858-5 also covers the design of fastener shock-resistant attachment/isolation of equipment/structures, penetration protection (access/egress or umbilical), hydraulic surge protection (circuits, reservoirs), and EMP protection.

c. TM 5-858-7 presents design guidelines for the facility support systems: power supply, waste-heat rejection, air quality control, utilities, and services. Each one of these has a direct impact on HVAC design which cannot be considered in total isolation but must be integrated in the total system-engineering approach.

(1) As an example, consider the total parasitic load which includes power for the coolant circulation pump, the refrigeration systems (largely compressors), and the air circulation fans. The parasitic load will often be dominated by fan power demand. As a result, at least a conceptual design of the HVAC is required to size the newer supply.

(2) The size of, the power supply in turn determines the power cycle heat rejection and the combustion air requirements. Both of these elements are part of the HVAC design, which must now be reevaluated to include these loads.

d. The air-conditioning (AC) of hardened ‘structures aboveground is essentially designed like conventional AC to hold the interior temperature, relative humidity (RH), and air supply at levels and volumes suitable for the intended use of the space.

(1) Underground this holding phase is preceded by a so called conversion phase, due to the much longer time interval required to warm up or cool down the initial temperature and RH of the underground space to the desired levels. The process of conversion must include the simultaneous control of temperature and humidity. Neither the addition or extraction of heat alone, nor the use of ventilating air alone, will ordinarily be sufficient for conversion purposes within acceptable time frames.

(2) The latent dehumidification load is usually greater than the sensible load during conversion; however, the sensible heat rejected by the dehumidifiers will be reused to heat the space, except for refrigerated storage cool-down below initial temperature levels. During conversion the structure will not be used for either production or storage, except in cases of emergency.

e. In hardened structures, ventilation alone will not suffice since dissipating the heat with outside air quickly becomes impractical. Therefore, a minimum quantity of outside air will be introduced with provision for complete recirculation and some degree of AC to provide for a greater latitude in occupancy and operational loads. During the seal-up period, the recirculation and cooling of interior air will permit continued operation and occupancy that may otherwise be prohibited. AC systems will be kept simple and designed for minimum maintenance.

2-2. Makeup air.

a. The proper quantities of outside air required for personnel are determined by pressurization, air lock scavenging, occupant metabolism, and other special requirements, such as for smoke purge systems.

(1) Leakage of underground structures is inexistent, and above-ground a gastight enclosure is required to prevent air contaminants from infiltrating the facility under attack. Air lock scavenging requirements (discussed in chapter 6) are proportional to the time allocated to personnel ingress. Fresh air provisions for personnel support are to dilute body odors, tobacco smoke, cooking, and other products due to occupancy.

(2) The American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE) Standard 62 lists minimum and recommended ventilation rates for various residential, commercial, industrial, and institutional structures. The normal allocation is 10 cubic feet per minute (cfm) per person for nonsmoking areas and 15 cfm per person for smoking areas. The lower limit for outside air ventilation is 5 cfm per person for maintaining proper carbon dioxide and oxygen levels.

b. In general it will be more effective to use an open ventilation system during the pre-attack time frame. The design of this system is similar to the ventilation systems found in conventional facilities, except that a hardened AES will be used to exchange air between the facility and the atmosphere. The AES is discussed in TM 5-858-5. Transattack/post-attack ventilation systems that communicate with the atmosphere must provide for the removal of insidious DB and other contaminants. This extremely difficult task will be avoided by using a closed ventilation system whenever possible.

c. Mechanical ventilation of underground installations is a necessity because natural ventilation is not practical for enclosed structures of facilities such as kitchens, dining areas, and lavatories. During normal periods of operation, there will be no recirculation of air supplied to kitchens, lavatories, toilet rooms, bathrooms, restrooms, and battery rooms. During seal-up, this air will be recirculated through carbon filters for odor removal. Recirculation of battery room air is permissible if batter charging operations are interrupted; otherwise hydrogen scrubbing will be provided. Air from decontamination areas will not be recirculated unless first passed through CB filters.

d. The ASHRAE recommendations for kitchens is at least 8 air changes per hour, but no less than 4 cfm per square foot of floor area. The quantities exhausted through hoods over ranges and other cooking devices will be sufficient to maintain a velocity of 60 to 75 feet per minute (fpm) through the projected area. For lavatories, toilet rooms, bathrooms, locker room, and restrooms, at least 4 air changes per hour but no less that 200 CFM, 7 cfm per locker, 25 cfm per water closet or urinal, 50 cfm per showerhead, or 2 cfm per square foot of floor area, whichever is greater.

e. Air supplied to offices and workrooms and exhausted via corridors will be used to ventilate toilets. For battery rooms, at least 1 cfm per charging ampere will be provided, but no less than 6 air changes per hour. Specific system applications covered in TM-5-810-1 are; administrative, community, storage, and computer facilities; research and development laboratories; and mechanical rooms.

2-3. Combustion air.

a. The proper quantities of outside air required for combustion processes are determined by the power supply and heat generators. Design guidelines for power supply are covered in TM-5-858-7. The diesel engine is the most likely prime mover for the power systems. Batteries and similar energy storage systems, which are bulky and have limited capacity, are practical only for the smaller shelters. Geothermal and nuclear-reactor-based power systems will only apply to the very largest facilities.

b. Space limitations and exorbitant combustion air requirements of coal and oil fired boiler plants for power generation all but eliminate these from consideration, especially in underground installations where combustion type boilers are excluded. For gasoline and diesel engine drives, air requirements per kilowatt (kW) are in the 4 to 7 cfm per kW range. For gas turbine drives, the range is from 9 to 13 cfm per kW.

c. Because outside air for personnel may be interrupted during the attack mode, combustion air for power generation must come from the structure itself or from an air intake structure separate from the air intake for personnel. Steam and hot water boilers may be used for heating and domestic hot water loads. These loads are usually expressed in British thermal units per hour (Btuh). For boiler combustion air estimating purposes, use 2.5 cfm per 10,000 Btuh. This combustion air requirement maybe combined with the personnel outside air requirements.

d. If a facility with an air-breathing power plant must be totally hardened, and if power production is required before, during, and after an attack, the air intake and exhaust equipment must remain operational at all times. The combustion air system will then include provisions to moderate the air inlet temperature and filter and scrub all dust and crater ejects from the prime mover combustion air.

(1) A typical installation would consist of a dry inertial dust separator to remove approximately 90 percent of dust particles 20 microns and larger, and a wet scrubber to remove 85 percent of dust particles 5 to 20 microns and to reduce the air inlet temperature in degrees Fahrenheit (°F) to less that 150° F.

Normally, the engine will be equipped with a viscous impingement-type filter in the combustion air intake for normal operations.

(2) Design of hardened ports and combustion air duct work is covered in TM-5-858-5.

e. The location of combustion engines and other air-consuming equipment is also of primary importance. Such equipment will be provided with a closed system with its own filtered air, or so located within a structure that the filtered air required for personnel will exhaust through the equipment area and be used for combustion. In an occupied structure without benefit of fresh air, any equipment requiring air for combustion will soon create an untenable condition within the structure. Such equipment will be isolated and provided with its own air supply and exhaust.

2-4. Vitiated air.

a. A concentration of 0.5 percent carbon monoxide in the air can cause death after one hour. The gas from a high explosive bomb can contain from 60 to 70 percent carbon monoxide. The air intakes and exhausts of a facility under attack will be sealed to protect against any such weapon effects. The length of time a facility must remain sealed up in the attack mode without rejuvenation of air will be determined during criteria development. Limiting parameters are temperature and humidity rise, oxygen depletion, and carbon dioxide buildup. These factors reviewed below are further discussed in the ASHRAE Handbook, Application.

b. The temperature and humidity rise in occupied unventilated spaces may be estimated by the methods of chapter 3. The particular case of an underground facility isolated for one week is considered in problem 5, paragraph 3-9e, using sedentary personnel metabolic emission rates (shown in table 2-1) as the sole source of heat build up in the space. Depending on initial and boundary conditions it is estimated that during the isolation period personnel will be exposed to temperature of 80° F to 90° F with humidities approaching 100 percent. This is not beyond human endurance, but is beyond the range at which work with paper, instruments, or electronic equipment can be reliably accomplished.

c. The hourly oxygen depletion rate under perfect mixing conditions is the ratio of the individual oxygen consumption V_{02} in cubic feet per hour (cfh) to the space volume per capita V in cubic feet. As a result, after t hours (h) of isolation, the oxygen volume fraction $[0_2]$ drops from the initial 21 percent normally present in the air to

$$[0_2] = 0.21 - t(V'_{02}/V_c) \quad \text{(eq 2-1)}$$

d. The hourly carbon dioxide buildup rate under perfect mixing conditions is the ration of the individual carbon dioxide production V' in cfh to V. As a result, after t hours of isolation, the carbon dioxide volume fraction [CO] will rise from the initial 0.3 percent normally present in the air to

$$[CO_2] = .003 + t(V'_{02}/V_c) \quad \text{(eq 2-2)}$$

e. Table 2-1 shows the various quantities of air, oxygen, and carbon dioxide used or given off under various conditions.

(1) Variation of oxygen levels between the normal 21 percent and 17 percent are acceptable, but carbon dioxide buildup is more serious because it acts on the human nervous system to maintain involuntary respiration. TM 5-858-7 indicates that hyperventilation and increased oxygen consumption will start above the 1 percent level and that carbon dioxide concentrations higher than about 4 percent are toxic.

(2) If in the problem of b above, V_c = 1,500 cubic feet per seated occupant each releasing 0.67 cfh of carbon dioxide, then equation 2-2 shows that the 4 percent critical carbon dioxide level is reached after only 90 hours, at which point the oxygen consumed at the faster rate of 0.8 cfh has also dropped (equation 2-1) below the 17 percent acceptable oxygen level. In other words, the occupants of the shelter will die asphyxiated long before their scheduled rescue (after 192 hours of isolation) unless air-regeneration processes are used.

f. A number of materials for chemically rejuvenating the air are given in table 2-2. These materials are useful for relatively small capacity carbon dioxide removal requirements. These regeneration processes also liberate heat and moisture as indicated. Their contribution to the latent and sensible heat load will be taken into account in the design.

(1) As a rule oxygen will be provided under pressure in bottles, but small quantities of oxygen may be generated by burning special chlorate candles. Oxygen to be generated or released can be estimated based on 0.89 cfh per person.

(2) For large-capacity carbon dioxide removal, counter current wet scrubbing with a sodium hydroxide solution is recommended in TM5-858-7, on the basis of reactant and low heat of reaction.

(3) Further discussion of air quality control is also covered in of TM 5-858-7.

TABLE 2-1

Typical Personnel Metabolic Rates

Physical Activity	Energy Expenditure, Btu/h	Oxygen Consumption, cfh	Carbon Dioxide Production, cfh	Rate of Breathing cfh
Prone, at rest	300	0.60	0.50	15
Seated, sedentary	400	0.80	0.67	20
Standing, strolling	600	1.20	1.00	30
Walking, 3 mph	1000	2.00	1.67	50
Heavy work	1500	3.00	2.50	75

TABLE 2-2

Properties of Air Regeneration Chemicals

Chemical[5]	Carbon Dioxide Absorbed	Oxygen Liberated	Water Vapor Freed	Sensible Heat Load	Latent Heat Load	Chemical Use Per man hour[6]
	ft^3/lb	ft^3/lb	lb/lb	Btu/lb	Btu/lb	lb/mh
Lithium[1] Hydroxide	6.74	N/A	.375	1,011	482	.124
Cardoxide[2] (soda-lime)	2.82	N/A	.240	380	308	.340
Baralyme	1.56	N/A	.116	200	150	.474
Sodium Superoxide	2.88	3.94	N/A	1,190	N/A	.288
Potassium[3] Tetroxide	2.28	3.27	N/A	815	N/A	.364
Chlorate[4] Candle	N/A	4.15	N/A	390	N/A	.240

1. 21.6 lb canister with 80 cfm blower @ 2.5 in. wg will last 2 hours
2. Irritating dust
3. 15 lb canister requires 50 cfm blower and will last 1.5 hours
4. Cast block 4-3/4 in. dia. 10 in. long sodium chlorate, iron powder, glass fiber, and barium peroxide must last 1 hour
5. These chemicals, if used, should be handled and stored with special caution. In particular, sodium superoxide and potassium textroxide are strong oxidizing agents and can be fire hazards. This fact may preclude their use in some cases. Chlorate candles should come packaged specifically to avoid fire hazard.
6. Based on 1 cfh O_2 and 0.83 cfh CO_2 per capita and forced air flow through the scrubbing chemicals (except chlorate candle).
7. N/A = not applicable.

2-5. Temperature and humidity.

a. Temperature and humidity of an occupied space have a direct effect on the occupants. A comfortable environment is essential for personnel who perform duties which demand reliable judgement and mental or physical stamina. Psychological stresses are likely to be present in individuals stationed in an underground structure under attack or alert. Where personnel efficiency is the prime consideration, TM 5-858-7 recommends 73 °F ± 1 °F effective air temperature, 80 °F + 2 °F dry-bulb air temperature, and 50 percent RH (optimum for control of air-borne bacteria). For additional guidelines refer to TM 5-810-1, and ASHRAE Handbooks.

b. Where operating equipment is the important consideration (electric racks, gyroscopic and celestial navigation equipment, laser missile tracking devices, and other similar equipment) temperature, humidity, flow, pressure, cleanliness, and other cooling air requirements will be designed in accordance with the equipment manufacturer's recommendations.

c. Information on the relation between humidity and deterioration of stored materials is shown in table 2-3. The data indicated the necessity for a low humidity for the preservation of unprotected carbon steel. As a result of these tests and other consideration, a RH of 35 percent was chosen for the interior of many ships place in storage. The 35 percent figure is considerably below the demonstrated tolerance of many materials, but it affords a factor of safety against equipment failure and against sharp temperature changes that might cause condensation on some objects when the temperature is uncontrolled.

d. Excessive dryness is harmful to certain materials, such as commutator brushes in electric motors, paper, excelsior, straw, leather, hemp rope, wood furniture, and dry-cell batteries. Recommended air conditions for storage of propellants will not exceed 60 percent RH with dry bulb temperature kept between 50 "F and 600 F. The AC equipment for explosive storage chambers will be selected with reference to minimizing pipe and duct runs. Where human occupancy is infrequent, little or no ventilation will be required. Equipment capable only of dehumidifying and moderately heating such chambers may be adequate in such situations.

TABLE 2-3

Humidity Tolerance of Selected Materials

Item	A	B	C	Nature of Damage
Mild steel, polished, unprotected	15	30	65	Rust
Steel (ball bearings, rust preventive applied by manufacturer)	--	65	90	Rust
Steel (ball bearings, heavy polar composition)	--	65	--	
Alloy steel	--	90	--	
Galvanized steel	--	65	90	Tarnish and rust
Brass and Bronze	15+	90+	--	Tarnish
Aluminum and its alloys	--	90+	--	Tarnish
Rubber, plastic, rayon	--	90+	--	Mildew
Flax, wool, cotton, hair, leather, sponge, hemp, sisal, paper, wood	--	65	90	Mildew
Soap, bars	--	--	90	Disintegration
Tinned cans (canned food)	--	45	--	
Cloth (life preserver)	--	65	90	Rotting of cover
Paint brushes	--	65	--	
Small arms, lubricated	--	65	90	Mildew and rust
Instruments (clocks, gages, voltmeters, telescope, etc.)	--	45	--	

Deterioration at indicated percent RH and uncontrolled dry-bulb temperature after a 30-month period; A = imperceptible, B = Very slight, C = Intolerable

U.S. Naval Industrial Test Laboratory Report 3014A, Philadelphia Navy Yard (April 1949)

2-6. Internal loads.

a. Electric motor driven equipment. The heat equivalent of one horsepower (hp) is 2545 Btuh, and a machine rated at K brake horsepower (bhp) dissipated heat at the rate q_e in Btuh.

$$q_e = 2545 (K) \qquad \text{(eq 2-3)}$$

This power, delivered by the motor, is a fraction of the motor input. This fraction is by definition the full load efficiency E of the motor and the motor input equivalent q_i in Btuh is then

$$q_i = 2545 (K/E) \qquad \text{(eq 2-4)}$$

The heat dissipated by the motor alone is the balance between motor and shaft input $q_m = q_i - q_e$ and in Btuh.

$$q_m = 2545(K/E)(1\text{-}E) \qquad \text{(eq 2-5)}$$

The efficiency of fractional horsepower motors increases with the rated power from a low 35 percent to a maximum of 76 percent. For estimating purposes the heat emitted by fractional horsepower motors is:

$$q_m = 800(K)^{.31} \qquad \text{(eq 2-6)}$$

From 1 to 250 hp, the efficiency continues to improve to reach a maximum of 94 percent. The approximate heat emitted by motors in that range is:

$$q_m = 800(K)^{.78} \qquad \text{(eq 2-7)}$$

The heat emitted by the motor or the driven machine must be allocated to the spaces or air stream where they are respectively located. If the shaft goes through a partition, these spaces may not be the same. The preceding calculations are valid for continuous operation. For intermittent operation an appropriate usage factor should be used and preferable measured.

b. Lights. Energy from electric lights is converted into heat. The heat equivalent of a watt (W) is 3.41 Btuh. The instantaneous lamp heat emission is the product of the heat equivalent of the total lamp wattage and a use factor which is the ratio of the wattage in use to the wattage installed. To this must be added the heat radiated by the ballast, which is usually 20 percent of the lamp heat emission. It bears repeating that the heat emitted by the fixture and the ballast must be allocated to the respective space or air stream where the heat is actually radiated. These spaces are not the same in the case of a false ceiling or when the fixtures are recessed or used for air return, and only a portion of the lamp heat reached the room it lights. This information should be supplied by the fixture manufacturer.

c. Occupant. Personnel emit sensible and latent (moisture) heat in the room. The individual emission rates depend on clothing, activity level, sex, age, room temperature T, and other factors influencing the person's metabolism. On the average the sum total q_t of the sensible and latent heat emission rate per capita is 400 Btuh for sedentary activities and 660 Btuh for light work. For design purposes the sensible heat rate q_s is 320 Btuh up to 68 F ambient, zero above 100 F, and between these limits computed according to equation 2-8.

$$q_s = (10)(100\text{-}T) \qquad \text{(eq 2-8)}$$

The latent heat rate q_l is by definition the balance between the total and sensible heat rate above or:

$$q_l = q_t - q_s \quad \text{(eq 2-9)}$$

For other condition the ASHRAE Handbook, Fundamentals, must be consulted.

d. Kitchen. Cooking is responsible for both sensible and latent loads. Appliance surfaces contribute most of the heat in kitchens in the form of radiant energy. Appliance heat loads are detailed in ASHRAE, Fundamentals.

(1) If the appliance is under an exhaust hood, the maximum heat released to the kitchen due to radiation is estimated at 32 percent of the rated heat input. With a 50 percent factor for diversity and the effect of thermostatic controls, the average heat emission in the room is then 16 percent of nameplate rating in Btuh, and the balance or 34 percent goes to the hood exhaust stream.

(2) For direct fuel fired appliances, a correction factor must be applied because they require 60 percent more heat input than electric or stream equipment of the same type and size, and the heat radiated in the kitchen is only 10 percent of the rated Btuh input.

(3) For all cooking appliances not installed under an exhaust hood, the heat gain maybe estimated at 50 percent of the rated input regardless of the type of energy or fuel used. It may be assumed that 34 percent of the heat is latent and 66 percent is sensible.

e. Engines. For diesel and gasoline engines, the only heat gain to be considered as internal load is the radiated load, estimated at 370 to 400 Btuh per bhp. The heat transferred to the cooling water is covered in chapter 5. For boilers, the heat radiated intro the conditioned space will be dependent on the temperature difference between the interior of the boiler and the conditioned space, the overall coefficient of heat transfer of the boiler plate and insulation, and the surface area of the boiler. This heat gain will have to be calculated for the specific boiler selected. Proper selection of insulation can reduce this load to a minimal amount.

f. Miscellaneous equipment. Other equipment using power such as computers, radar, and communication equipment will also produce a heat gain based on its specific characteristics as indicated by the manufacturer.

2-7. External load.

a. The sensible and latent heat transfer between the space and its surroundings constitute the external load. Aboveground, the ambient air is the dominant factor. Solar radiation's influence is always indirect because of the absence of fenestration in hardened structure. In addition, the thickness of the walls will reduce considerably the propagation of the external daily temperature variations to the inside spaces.

b. Moisture seepage through boundaries aboveground will be eliminated just as it is for conventional structures; however, the designer should be aware that the thermal parameters of the overburden used in mounded-over structures, for instance, are sensitive to moisture content and therefore dependent on local precipitations, regardless of the drainage and moisture barrier provided.

c. Underground, the structure's environment will, by contrast, vary in texture, contain fissures or faults, and be subject to hidden hydrostatic and thermal influences. Heat transfer from this environment is covered in chapter 3. Temperature variations in the underground environment are relatively small over a period of time compared to the wide ranges of seasonal variations which affect an aboveground structure.

d. The intrusion of moisture into the underground structure is much more unpredictable and difficult either to measure or control than in the aboveground structure. Therefore, the design of the AC system for the underground structure will anticipate relatively constant temperature levels but varying moisture conditions. Each site will present a unique series of geologic and geographic conditions. No uniform design procedure can be applied universally to all sites.

e. The underground environment is exceptionally dominated by the movement of groundwater around the space; but the usual assumption is that this percolation will not eliminate conduction from consideration and that both transfer mechanisms can be evaluated separately.

2-8. Moisture loads.

a. Evaporation of water from damp surfaces or open pools into the air requires heat. At normal room conditions the latent heat of evaporation is approximately 1050 Btu per pound of evaporated water. The latent heat of evaporation is transferred by the resulting vapors from the wet interface to the place where the vapors recondense. Water vapor in the air will recondense on any exposed surface at a temperature below the air dewpoint.

b. The vapor pressure P_w of the water, in pounds per square inch absolute (psia), increases rapidly with the temperature T in 1? according to the Tentens formula.

$$P_w = (,0886)\exp[(17.2694)(T-32)/(395.14+T)] \qquad \text{(eq 2-10)}$$

The saturated vapor pressure P_s of air at dry-bulb temperature T_a is computed by setting $T = T_a$ in equation 2-10. The actual vapor pressure P_a of the air is then computed from its known relative humidity which, by definition, is the ration of P_a to P_s. The air dewpoint temperature T_b is also computed from equation 2-10 by setting $P_w = P_a$ and using parameter $P^* = 1n\,(P_a/.0886)$ as follows:

$$T_b = (395.14)(1.3985+P^*)/(17.2694-P^*) \qquad \text{(eq 2-11)}$$

c. **The driving force for evaporation from a wet surface is the saturation deficit ($P_s - P_a$), which is** positive above the air dewpoint. Air movement is also a factor since it prevents vapor buildup and saturation of the air above the evaporating surface. For a surface L feet long with air flowing parallel to it at a velocity v in fpm, the average mass transfer coefficient in lb/h ft^2 per psi saturation deficit is approximately:

$$G_p = (v/1538) + (.22/L)\,[1-\exp(-vL/135)] \qquad \text{(eq 2-12)}$$

The mass flux m in 1b/h ft^2 of water evaporated is then:

$$m = G_p\,(P_s - P_a) \qquad \text{(eq 2-13)}$$

The latent heat flux in Btuh/ft^2 corresponding to m is approximately:

$$q = 1050(m) \qquad \text{(eq 2-14)}$$

For more complicated interface geometries, G_p is computed by analogy with the heat transfer coefficient h' in Btuh/ft^2 F, using the Lewis relation for moist air at 14.7 psi absolute pressure, 0.24 Btu/lb F specific heat, and 0.622 water to air molecular weight ratio.

$$G_p = (.622/14.7)\,(h'/.24) = .18(h') \qquad \text{(eq 2-15)}$$

d. Underground, the computation of moisture loads will be based on site specific data. A site survey will determine the amount of water entering trough fissures, collected in pools, and the excess water to be drained or pumped away. Determining rock seepage and other hidden or intermittent sources will require extended observations. Moisture from equipment, materials, processes, personnel, fresh air, infiltration from uncontrolled areas, and other sources will be included in the design. Experience indicated that failure to account for these loads disrupts the entire humidity control process.

e. Evaporation from damp rock affects the humidity in bare chambers. Initially, when the chamber is first warmed, the rock surface is below the air dewpoint. Moisture from the air condenses on the rock, adds to the existing seepage, and reduces the room latent load. The latent heat of condensation released adds to the sensible heat flux penetrating the rock. Upon continued heating, the rock surface temperature rises. When the surface temperature is above the air dewpoint, part of the air sensible heat is converted and used to evaporate some of the seepage at the rock surface, thus reducing the heat flux penetrating the rock. The moisture added to the air increases the room latent load.

f. Vapor barriers or thermal insulating materials in direct contact with rock surrounding underground spaces are not generally recommended.

(1) Hydrostatic pressures generated because of the depth of an underground chamber are greater than can be restrained by ordinary vapor barrier materials or even by moderately heavy concrete liners. Hydrostatic heads up to 43 psi could develop 100 feet below the water table depending on the over-burden permeability.

(2) Insulating material applied directly to rock walls or to concrete in contact with such walls is likely to become wet, either by condensation or from ground water or both, with possible damage to the insulating materials or to the fastenings.

(3) A concrete liner may be installed in an underground space to improve the appearance or to reduce the chances of spalling, but should not be considered effective either as thermal insulation or as a vapor barrier. The dehumidification load in such a space is the same as that for a bare chamber.

g. If the walls, ceiling, and floor of an internal structure are vapor proof, the water vapor to be removed is equal to that liberated by the equipment and personnel within the structure. Conditions in the annular space do not directly affect conditions within the structure. If the walls, ceiling, and floor of the internal structure are pervious, the water vapor to be removed is the sum of the water vapor liberated by personnel and equipment and that entering the internal structure through the walls, ceiling, and floor by permeation or by convection from the annular space.

2-9. Air distribution and fire protection.

a. General configuratwn.

(1) A central AC! system has the advantage of lower chilled and hot water piping first-cost and lower noise generation when the unit is remote from the conditioned spaces. Disadvantages are large, long ducts, inflexibility under moderate load, and the inherent unreliability of a single system when compared to installation of multiple units.

(2) On the other hand, using a large number of self-contained air-conditioners , one for each room or zone, simplifies the zoning and control problems, improves the overall reliability, and avoids the use of large, long insulated ducts. Noise may be a problem if such equipment is used because human occupants may be situated near the source of the noise. Self-contained air-conditioners include condensing units in pre-assembles cases. For underground use, these condensers will be water cooled.

(3) Fresh air will be ducted to the return side of the air-conditioner in proportion to the population in the room or zone being conditioned. This allows the air to be tempered or bypassed through a conditioning coil before entering the occupied space..

(4) Self-contained air-conditioners will be furnished with hot water or steam coils when heating is required or arranged to serve as heat pumps and, thus, warm as well as cool and dehumidify spaces when required. Most of the heat for warming a space with a heat pump arrangement is taken from the water used at other times to cool the condenser.

b. Distribution. Prior to design of the air distribution system, the designer of HVAC systems for multi-room structures will analyze the requirements for each room. Areas containing odors, toxic vapors, dust, and other contaminants will be designated as contaminant areas. All other areas will be designated as non-contaminant areas.

(1) Contaminant areas will be maintained at a lower pressure relative to adjacent rooms to ensure that contaminants generated within the area will not escape to other areas. To obtain the maximum utilization of ventilating air, exhausts from toilets, and kitchens (properly degreased) will be discharged into unoccupied equipment rooms.

(2) Filtered air will be distributed in a manner to give the most effective results in providing uniform air quality for occupants. Filtration requirements will be specified as a function of the location of the facility and the air quality required to accomplish its mission. A duct system will be used wherever feasible, except between areas or rooms where pressure differences are to be maintained.

(3) In structures not provided with central AC or air-handling equipment, circulation or recirculation of air can be obtained by the proper placement of floor or wall fans. During seal-up periods

when fresh filtered air cannot be supplied, this continued recirculation of air will extend the time of occupancy.

c. Air motion. Air motion in comfort air-conditioned spaces should be at a design rate of 50 fpm. In areas where people will be seated, such as in offices, control rooms, and personnel support areas, room air motion should be ,between 25 and 35 fpm.

d. Fire protection. The designer of HVAC systems for hardened structures will take special precautions to prevent the spread of fires through ducts and shafts.

(1) Fire dampers and smoke detectors will be installed in all duct systems in accordance with National Fire Protection Association (NFPA) publication 90A. In the event of a fire, flow of smoke from the fire zone will be inhibited from spreading to required interior ways of exit access, interior enclosed stirs and ramps, passageways, and designated refuge areas.

(2) Smoke control and purge systems will be included as an integral part of the HVAC systems. Such systems will involve HVAC systems alone or in combination with other systems such as emergency venting, pressurizing systems, and fire suppression systems and are covered in the ASHRAE Design of Smoke Control Systems for Buildings.

e. Radio frequency interference (RFI) protection.

(1) Supply and return air ducts serving rooms with sensitive electronic equipment will be equipped with RFI filters to prevent transmission of RFI into the electronic equipment room. This RFI protection requirement is usually included in the design criteria.

(2) To maintain the attenuation of the RFI shielded room at the prescribed level, a waveguide filter will be installed in the duct which will result in attenuation equal to the center area attenuation of the room. The air-duct wave guide filter will be specified in terms of the attenuation over a specified range of RFI frequencies and the allowable air pressure drop across the filter in accordance with TM 5-855-5 and TM 5-858-5.

f. Internal structures.

(1) To conserve space an internal structure will be cooled by utilizing the annular space between the structure and surrounding rock as a cold-air plenum held at or near the initial rock temperature. The cool air in the annular space plenum is distributed into the internal structure to maintain the desired interior conditions.

(2) Air from the internal structure, except for that exhausted, will be discharged directly into the annular space through a cooling coil that lowers the air temperature to that of the surrounding rock. Leakage of air between the plenum and access passageways will be prevented to avoid discharge of cold air into warm passageways.

2-10. Chemical, biological, and radiological (CBR) protection.

a. General. Protection against CB agents and radiological fallout will be provided if the facility is to continue to function regardless of attack. There are not varying degrees of CBR protection, and it will be continuous if the effects of covert attack are to be reduced.

(1) Air filtration, pressurization, and personnel decontamination are the three basic principles of CBR protection. Radiation shielding and CBR detection are not part of this manual. Entrances and decontamination facilities (covered in chapter 6) will permit egress and ingress without endangering the occupants of the facility.

(2) CBR agents may occur as gases, liquids, or solids and can be airborne, waterborne, or surface contaminants. Airborne agents are chemical toxic gases and CB aerosols. Surface contaminants are too heavy to remain suspended in the air. They can be either liquids or solid. Liquids may be chemical or biological agents. Solid contaminants may be biological or radiological agents.

b. Pressurization.

(1) Dependable exclusion of airborne agents is ensured by sealing possible leakage sources, providing pressurization, and filtering the makeup air. The internal overpressure P in inches of water (in. wg) needed to prevent infiltration from winds of velocity v in miles per hour (mph) is 110 percent of the velocity head equivalent of the wind or

$$P' = 0.85(v/40)^2 \qquad \text{(eq 2-16)}$$

(2) Air input needed to achieve the desired overpressure is determined by the exhaust and leakage rate of the installation and is independent of the size of the installation. Contaminants are effectively removed from air by passing them through a standard filter unit developed by AMCCOM-CRDC. After

passing through the filter, the air is collected and distributed within the installation by means of a supply fan and a suitable duct system. Air distribution strategies given in the preceding paragraph will maximize its utilization.

(3) Control of exhaust air and sealing air leaks will provide a degree of pressurization. Exhaust air control is achieved by poppet valves or other valves calibrated to permit a fixed flow of air under pressure or by volume control dampers on exhaust fans. The valves will be located as far as practicable from the fresh air intake to provide a good circulation of air through the installation. Exhaust air control of entrances is discussed in chapter 6.

c. Air scrubbing of openings. Many CB contaminants will tend to concentrate in subsurface openings such as shafts and cut entrances. Such openings will be sealed off at or above ground level. For some kinds of openings, such sealing will be impossible. To prevent or reduce the build-up of contamination, provision should be made to scrub the openings with exhaust air from the tunnel or by other means. Scavenging air of decontamination facilities are discussed in chapter 6.

d. Exclusion of solid and surface contaminants.

(1) Surface contaminants can enter an underground installation either by falling into openings or, after they have been deposited on the ground, by being carried in accidentally by personnel or vehicles. Shielding over openings will prevent such contaminants from falling into the installation. Offsets in shafts are not enough protection against this danger, for they require that the contaminants be removed or decontaminated, and until decontamination there is danger that the contamination may be moved on into the installation.

(2) Dust removal devices are covered in TM 5-858-5, but ease of removal and disposal of contaminated media must be considered in the equipment selection. In this respect, dry type traveling curtain air filters are preferred to fabric-bag dust separators.

(3) Prevention of contamination by surface contaminants carried into the installation is a matter of detection and decontamination. Decontamination is covered in chapter 6.

e. Air intakes.

(1) At least two air intakes will be provided whenever possible and given a maximum separation to reduce the possibility of both intakes being destroyed by a single explosion. Each intake shaft will be capable of handling the total air requirement of the installation with a minimum friction loss and will extend above the structure or earth in such manner as to preclude areas of possible high concentration of contamination.

(2) Air intake cross sections will vary from 1 to 5 square feet with the larger shafts used also as normal air intake for ventilation and AC of occupied areas. For these areas, the incoming air must pass through chemical filters prior to entering the distribution or conditioning equipment. Bypass of these CB filters will not be allowed.

(3) Should it be necessary to duct contaminated air through protected areas within a structure, it is essential that the internal duct pressure be less than that of the area through which it runs. This condition will permit an inward flow of air into the duct in the event of a leak. Design of intake ports is covered in TM 5-858-5.

f. CB filters.

(1) The filter units developed by AMCCOM-CRDC, described herein are a development of Aberdeen Proving Ground, will be installed in all military structures that are to be provided with CBR protection. HVAC designer will specify component particulate and gas filters, only as listed in table 2-4 and 2-5. When properly installed, these filter units will provide maximum protection against CBR contaminants that may enter a structure through the ventilation air intake.

(2) The CB filters are composed of two units in series, one unit being a dense water-repellent paper for the retention of particulate matter, which is the basic carrier of biological and radiological contaminants. The second unit is an activated carbon unit for the adsorption, retention, and neutralization of chemical agents.

(3) As no neutralization of the biological and radiological contaminants can occur, the filter units may become a secondary hazard to personnel in the immediate vicinity of the filter units. The construction of the filters does not provide for the required radiation shielding. As a rule and for an airflow V' in cfm, the shield surface density in pounds per square foot (psf) will be within 20 percent of (12.6 $(V')^{.3}$]. Shielding of the filters and the operating procedure to protect personnel when removing contaminated filters will be coordinated with AMCCOM.

(4) Provisions will be made to transfer contaminated filters to the outside without moving through protected areas. Proper location and installation of CB filters must be coordinated with the design of the

hardened AES covered in TM 5-858-5. Any questions regarding the types of prefilters that should be used in a particular system will be referred to AMCCOM.

(5) The filter units will be installed in a readily accessible location and be provided with an overhead hoist for periodic removal and replacement. The filters will be located as close as possible to an exit and remote from the occupied portions of a structure. In structures not provided with blast protection, — the filter units will be located outside the structure in the vicinity of the main air intake.

(6) When filters are located outside a structure or in an otherwise contaminated area, the supply fan will be place on the influent side of the filters to preclude the infiltration of contamination in the event of system leakage. When filters are located inside a structure or in an otherwise clean or protected area, the supply fan will be place on the effluent side of the filters. This arrangement will eliminate the infiltration of contamination since any leakage will be that of clean air.

(7) All intake air will be filtered continuously unless the total requirements necessary for normal operations of a structure make such filtration uneconomical. Automatic CBR detection devices cannot be relied upon to put the CB filters on the line when bypassing these filters is allowed; this operation will have to be accomplished manually at the start of the alert and in advance of pending attack.

(8) Normally, a standby system of filter units identical to the main units will be installed for occasions when the filters become contaminated and require replacement and when such replacement cannot be accomplished by shutting down the fresh-air supply. In certain important structures, such as deep buried ones, the standby filters will be on a completely separate system of supply fans and intake shafts in case one system becomes inoperable as a result of equipment failure or air-intake shaft damage. In other instances the standby filters may be stored in readiness for replacement rather than being initially installed in the duct system. The methods of providing standby filter units will depend on the importance of the mission to be accomplished within the structure as determined by the using agency.

g. *Filter equipment room*. A separate area or room will be provided for the air-filtering equipment, and when AC is required, portions of the AC equipment will also be placed in this room. This room will be pressurized with clean air, and the filter units and fans will be arranged so that any leakage into the room will be that of clean air. The refrigerant compressors and evaporative condensers of any AC system will be placed outside the pressurized area in order to reduce heat buildup, filtered-air requirements, and possible refrigerant leakage.

h. *Protective closures.*

(1) Protective closures are required at air intake and exhaust openings, plumbing vents, or other openings to the atmosphere to prevent a pressure buildup within the facility greater than 2 psi above atmosphere pressure. Blast closures, valves, ducts, attenuation chambers, debris traps, penetration protection, and hydraulic surge protection are covered in TM 5-858-5.

(2) In view of the complications involved in design and the variations in requirements for protective closures in a single given structure, it is strongly recommended that the entire system of air intakes and exhausts, soil vents, boiler stacks, engine combustion intakes, and exhausts be designed to reduce the number and types of closures required. This will be accomplished by consolidating a variety of exhausts or intakes into a common plenum having its own protective closure.

TABLE 2-4

Particulate Filters

CRDC Model No.	Rated Capacity (cfm)	Dimensions H (in)	W (in)	L (in)	Net Weight (lb)	Pressure Drop (in. wg)	National Stock No.
C18R1	600	24	24	6-1/8	18	1.00	4240-901-8119
C19R1	1200	24	24	11-1/2	40	1.75	4240-901-8118
C30R1	2500	24	46-1/2	11-1/2	64	1.75	4240-901-8117
C20R1	5000	48	48	11-1/2	120	1.75	4240-901-8116
M20*	1250	24	24	11-1/2	40	1.00	4240-892-5369

*Fire Resistant

TABLE 2-5

Gas Filters

CRDC Model No.	Rated Capacity (cfm)	Dimensions H (in.)	W (in.)	L (in.)	Net Weight (lb)	Pressure Drop (in. wg)	National Stock No.
C22R1	600	25-1/2	25-1/2	29-1/4	275	1.25	4240-901-8115
C32R1	1200	25-1/2	25-1/2	51-5/8	530	1.25	4240-901-8114
C29R1	2500	25-1/2	48	51-5/8	1000	1.25	4240-901-8113
C23R1	5000	48	48	50-3/4	2100	1.25	4240-901-8112
FFU-17/E	1000	24	24	18	245	1.80	4240-176-9992

2-11. Economic factors.

a. Space utilization.

(1) The selection and operation of equipment within a hardened facility are governed primarily by requirements other than economy, especially underground. The economics of equipment selection and operation will be compromised where dictated by facility mission requirements. The HVAC designer will review the equipment configuration and space allocation to provide a familiarity with maximum utilization of excavated space, minimum consumption of energy, and optimum hardness design to ensure successful completion of the mission.

(2) Trade-offs must be made in efficiency and noise when using smaller ducts with higher velocities and small high capacity equipment such as fans, coils, and boilers. Noise will be kept within limits set by Occupational Safety and Health Administration (OSHA); however, for each space, and where necessary, such design consideration as grouping and isolation of equipment and noise attenuation will be provided for maximum utilization of space.

b. Economy of operation.

(1) Facilities designed for uninterrupted power have continuous operation prime movers. Waste heat from jacket water and engine exhaust will be recovered to heat the facility and domestic hot water. Lube oil heat recovery may also be practical. In a gas turbine cycle, the thermal efficiency is approximately 12 to 60 percent with the remainder of the fuel energy discharged in the exhaust or through radiation. A diesel engine rejects approximately 30 percent of the input fuel energy to the jacket water and 30 percent to the exhaust gases.

(2) Practically all the heat transferred to the engine jacket water can be utilized but exhaust heat recovery is limited to 300 F leaving gas temperature to prevent condensation of water vapor and acids in the exhaust piping. Depending on the initial gas temperature, approximately 50 to 60 percent of the available exhaust heat can be recovered. Heat recovery methods are covered in detail in the ASHRAE Handbook, Systems.

(3) In frigid and temperate climatic zones, air-to-air heat exchangers or heat pipes will be installed in outside air and return air ducts for sensible heat reclamation. Where conditions permit, duct-mounted rotary air desiccant wheels will be installed in air-conditioning exhaust and outside air ducts for latent and sensible heat recovery.

(4) Where facility hardness requirements and interior humidity design conditions permit, outside air will be used to cool the facility when ambient dry-bulb temperature is 640 F or lower.

(5) Where high-radiant, heat-producing equipment, such as ovens, furnaces, and infrared devices are to be installed, consideration will be given to isolating such equipment by the use of metal panels through which water at normal temperature is circulated to carry off this high heat, thus reducing the load on the air-conditioning system. Similarly, the selection of liquid coolant-type power units, having water jackets through which either water at normal temperatures or condenser water can be circulated, will reduce the load on the “air-conditioning system.

2-12. Survivability and reliability.

a. General. Survivability and reliability of hardened structures are discussed in TM 5-858-1. By way of illustration, some of the HVAC applications of these considerations are included in this manual. Heating and Air-conditioning equipment installed in hardened facilities will be of such design or otherwise protected to withstand the shock (ground motion) and overpressure effects of weapons. Experience with the ballistic missile programs has proven that standard air-conditioning equipment can be utilized in hardened facilities if properly designed and protected.

b. Redundancy of equipment.

(1) Systems requiring a high degree of reliability will include redundant units which will automatically start and maintain the load should the operating unit fail. The required degree of reliability is based on the function of the facility, allowable downtime for critical systems, type of facility operation (continuous or standby), type of system operation (remote or local), and degree of maintenance.

(2) Fans and pumps in critical HVAC systems will be installed in multiples of two or three. The degree of reliability will determine whether units will be installed in multiples of two with each unit designed to carry 100 percent of the load or in multiples of three with each unit designed to carry 50 percent of the load.

(3) Controls will be arranged to keep one of the units in near new condition, operating it only as required for maintenance. In some cases, bypasses for control valves will be required for AC reliability where single AC units are used. Remotely operated valves in critical fluid systems will require two valves in a series to ensure reliability of facility isolation during the button-up phase. Computer cooling

systems must have a high degree of reliability; therefore, redundant components on the control system are justified.

c. Survivability of equipment.

(1) The overriding facet of hardened facility design is survival of equipment and personnel to complete the mission for which it was designed. A detailed dynamic analysis will be made of the supporting structure and the magnitude of motion and acceleration established at the mounting points for each piece of HVAC equipment. This is covered in TM 5-858-4.

(2) Medium weight machinery such as pumps, condensers, and air-conditioners, weighing from 1,000 to 4,000 pounds, can sustain accelerations (expressed in units of g the acceleration due to gravity) of approximately 15 g without damage. Air blast pressure on fans will be limited to 5 psi to prevent damage due to the rapid acceleration offers. Light machinery of 1,000 pounds or less such as fans, and small motors can sustain accelerations of approximately 30 g without damage. Where accelerations exceed the allowable limit of equipment available, the equipment will be mounted on shock isolation platforms.

(3) The design will include, where feasible, certain features which will enhance the survivability of hard-mounted HVAC equipment. Double inlet fans and double suction pumps will withstand shock forces generated by ground motion with the fan wheel and impeller supported on both sides without board bearings. Conversely, single inlet fans and pumps with overhung wheels and impellers will not be used in hardened installations unless they are mounted on shock isolated platforms.